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Considerations of Several Real Effects in Pneumatic Pellet Injection Processes

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CONSIDERATIONS OF SEVERAL REAL EFFECTS IN PNEUMATIC PELLET INJECTION PROCESSES

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Abstract. Several real effects that take place in a pneumatic pellet injector are examined. These are the heat transfer between a high-temperature propellent gas and the metal wall of the injector, and the frictional loss between the propellent and wall.

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1. INTRODUCTION

Several real effects that take place in pneumatic pellet injection processes are examined. Heat transfer between a high-temperature propellent gas and its metal wall is so high that most of the gas enthalpy is estimated to be lost within a fraction of barrel length. Either the use of a heat-insulated wall with high-temperature gas at the entrance or a heated metal wall with non-heated gas at the entrance is suggested to improve the performance of next generation pellet injectors. Frictional losses between the propellent gas and wall could be significant for high-speed injectors, and simplified calculations for both simple single-stage and constant base pressure acceleration cases are suggested to take account of this effect. Friction between the pellet and wall can be neglected if a gas film were to exist between the pellet and wall.

The ideal theory of single-stage and constant base injection had been summarized years ago[1]. Based on frictionless and adiabatic assumptions the isentropic relations of unsteady one-dimensional flow could be derived analytically in several simple cases and numerical results had also been worked out for geometric, more complex configurations[1].

For a large pellet size (order of centimeters), where the early interest was in reentry simulation, these assumptions seemed relatively reasonable.

Recent experiments on injecting small cryogenic solid hydrogen or deuterium pellets into fusion plasma (with a pellet size of few millimeters) showed significant deviations from ideal theory[2,3]. Since friction and heat transfer are subjected to surface effects, the momentum and enthalpy of the gas are subjected to volume effects. As the surface-to-volume ratio increases as the pellet size decreases, the assumptions for ideal case originally accepted should be reexamined more carefully.

In this report, three real effects that usually occur in pneumatic pellet injection have been studied; some conclusions came out quite differently from those originally considered.

It should be noted that the following approaches give estimates on very complicated phenomena only. More experiments and theoretical works should be followed if some effects are really shown to be important.

2. FRICTION BETWEEN PELLET AND WALL

When an iced solid pellet at a cryogenic temperature is to be launched in the barrel by propellent gas, direct solid contact between the pellet and wall seems quite impossible. Instead a small clearance will always exist as the pellet evaporates. An ideal case could be imagined as in Figure 1. Here it is clear that the propellent gas will leak through the clearance by the pressure drop across the pellet. The pellet will be subjected to a viscous force from the gas instead of from the wall.

Since the clearance is usually much smaller than either the pellet diameter or its length. A planer fully-developed laminar flow with an average viscosity will be chosen as a simplified model of this problem. It can be treated actually as a generalized Couette flow[4] determined by the following equation:

$$\mu_f \frac{d^2v}{dy^2} = - \frac{p_1 - p_2}{\ell_p} = - \frac{\Delta p}{\ell_p} \qquad (1\text{-}1)$$

The solution is

$$\frac{v}{v_p} = \frac{y}{\delta} + \frac{\Delta p \ \delta^2}{2\mu_f \ \ell_p \ v_p} \frac{y}{\delta} \left(1 - \frac{y}{\delta}\right) \qquad (1\text{-}2)$$

with the boundary conditions: $y = 0$, $v = 0$, $y = \delta$, $v = v_p$.

When

$$N = \frac{\Delta p \ \delta^2}{2\mu_f \ \ell_p \ v_p} = 0$$

it is the simple Couette flow without pressure gradient, a drag force acts on the pellet ($\tau_f = (\mu_f \ du/dy)_w$). When $N = 1$, the velocity gradient and hence the viscous force vanishes at the pellet surface. When $N > 1$, it turns out that the viscous force acting on the pellet surface gives a favourite direction to push the pellet instead of pull it.

With typical parameters for D_2-pellet injection

$\Delta p = 10$ bar

$\mu_f = 2 \times 10^{-5}$ dyne-s/cm^2 (1000 K)

diameter of barrel $d_b = 0.4$ cm

length of barrel $\ell_p = 0.4$ cm

$\delta = d_p/2 \ (1 - d_p/d_b)$

$= 2 \times 10^{-3}$ cm (assume pellet diameter is 1% less than barrel diameter)

v_p 1000 m/sec

$$\tau_f/y{=}\delta = \mu_f \frac{v_p}{\delta} - \frac{\Delta p}{\ell_p} \frac{\delta}{2} = - 24 \times 10^3 \text{ dyne/cm}^2$$

(with $N = 25$)

The total push force on the pellet due to a frictional effect as compared with the pressure force is

$$\frac{F_\tau}{F_p} = \frac{|(\tau_f)_{y=\delta}| \cdot \pi d_p \ \ell_p}{\Delta p \cdot \frac{\pi}{4} \ d_p^2} = 0.01 \qquad (1\text{-}3)$$

It is shown through this typical case that the force due to friction gives a favoured direction but compared to the normal push force acting on the pellet it still could be neglected. The conclusion is that if a gas film were to exist between the pellet and wall, it will be possible to neglect the friction between the pellet and wall even if the velocity of pellet were to reach much higher velocities (say to 5 km/s).

The gas that leaks through the clearance as compared to the amount to be used to push the pellet could be estimated as follows:

$$\frac{G_L}{G_o} = \frac{\rho\ \Delta t \cdot \pi\ d_b \int_o^\delta v\ dy}{\rho\ \Delta t\ v_p\ \frac{\pi}{4}\ (d_p^2)}$$

$$= \frac{4\delta}{d_b}\left[\frac{1}{2} + \frac{\Delta p\ \delta^2}{12\mu_f\ \ell_p\ v_p}\right] = 9.3\% \qquad (1\text{-}4)$$

with the same parameters as above. It seems that gas leakage will be quite serious when the clearance gets larger, since G_L/G_o is proportional approximately to δ^3.

3. FRICTION BETWEEN PROPELLENT GAS AND WALL

Though friction between the pellet and wall probably can be neglected as shown in the last section, the seriousness of the frictional loss between the gas and wall is estimated as follows:

a. Constant base pressure acceleration case: (Figure 2b)

For pellet:

$$M_p \frac{dv_p}{dt} = p_b\ A_p \qquad (2\text{-}1)$$

If p_b could remain constant during the acceleration, then the acceleration also is constant.

$$\frac{dv_p}{dt} = a_p = \frac{p_b\ A_p}{M_p} = \text{const.} \qquad (2\text{-}2)$$

The basic principle of constant base pressure acceleration is that all the gas following the pellet will continue with the same velocity and acceleration as the pellet itself. By neglecting friction the ideal theory can be found in [1]; here a closed form solution is derived by considering the frictional effect.

The gas following the pellet is determined by

$$\rho_g \frac{dv_g}{dt} = - \frac{dp_g}{dL} - f\ \rho_g \frac{v_g^2}{2} \frac{1}{d} \quad , \qquad (2\text{-}3)$$

where f is the friction coefficient [4].
According to the principle of constant base pressure acceleration, keep

$$\frac{dv_g}{dt} = \frac{dv_p}{dt} = a_p = \text{const.} \qquad (2\text{-}4)$$

$$v_g = a_p\ t \qquad (2\text{-}5)$$

Equation (2-3) can be written as

$$dL = - \frac{dp_g}{\rho_g \cdot a_p} - f \frac{a_p t^2}{2d} \frac{dL}{dt} dt \qquad (2\text{-}6)$$

since $\frac{dL}{dt} = v_p = a_p t$, Eq. (2-6) can be integrated under the assumption of a constant average friction coefficient f and polytropic relation between pressure and density.

$$p_g = \text{const.} (\rho_g)^n \qquad (2\text{-}7)$$

$$f \frac{L^2}{2d} + L = \frac{p_b}{\rho_g a_p} \left(\frac{n}{n-1}\right) \quad \frac{p_{g,o}}{p_b}^{n-1/n} - 1 \qquad (2\text{-}8)$$

since

$$L = \frac{v_p^2}{2a_p}\ ,\ a_p = \frac{p_b A_p}{M_p} = \frac{p_b}{\rho_p \ell_p}$$

Eq. (2-8) can also be written as

$$\frac{p_{g,L=o}}{p_b} = 1 + \quad f\ \frac{\rho_p \ell_p}{\rho_b\ d}\ \frac{K}{8}\ \bar{v}_p^4 + \frac{1}{2}\ \bar{v}_p^2\ \ \frac{K(n-1)}{n}^{\ n/n-1} \qquad (2\text{-}9)$$

Here, $\bar{v}_p = v_p/C_b$, $C_b = (KRT_b)^{1/2}$. If $f' = f\ \dfrac{\rho_p \ell_p}{\rho_b\ d} = 0$,

$n = K$, Eq. (2-9) reduced to,

$$\frac{p_{g,L=0}}{p_b} = \left(1 + \frac{K-1}{2}\ \bar{v}_p^2\right)^{K/K-1} \qquad (2\text{-}10)$$

for the ideal case [1].

Several curves from Eq. (2-9) for static pressure programming to constant base pressure acceleration take account of the frictional loss (as shown in Figure 3).

b. Simple single-stage case (Figure 2A)

For single-stage pellet injection, the momentum equation for the pellet is

$$M_p\ \frac{dv}{dt} = p_b\ A_p \qquad (2\text{-}11)$$

p_b is now a variable, and the momentum equation for the gas just behind the pellet is

$$\frac{dv}{dt} = - \frac{1}{\rho}\frac{dp}{dx} - f\frac{v^2}{2d} \qquad (2\text{-}12)$$

For polytropic process, the pressure-density relationship is

$$\rho = \rho_o \left(\frac{p}{p_o}\right)^{1/n} \qquad (2\text{-}13)$$

and the sonic speed-pressure relationship is

$$\frac{C}{C_o} = \left(\frac{T}{T_o}\right)^{1/2} = \left(\frac{p}{p_o}\right)^{n-1/2n} \qquad (2\text{-}14)$$

With Eqs. (2-13) and (2-14), Eq. (2-12) can be written as

$$\frac{dv}{dt} = - \frac{1}{\rho c}\frac{dp}{dt} - f\frac{v^2}{2d}$$

$$= \frac{-\dfrac{p_o}{\rho_o c}\dfrac{M_p}{A_p}\dfrac{d^2v}{dt^2}}{\left(\dfrac{M_p}{A_p}\dfrac{dv}{dt}\right)^{n+1/2n}} - f\frac{v^2}{2d} \qquad (2\text{-}15)$$

For an arbitrary polytropic process it is impossible to get an analytic solution from Eq. (2-15) (although numerical results could be obtained, of course). Fortunately, for the isothermal case (n=1). An analytical first integration could be obtained as follows: for n=1 ($c = (RT)^{1/2}$ = const.), Eq. (2-15) is reduced to

$$\frac{d^2v}{dt^2} + \frac{1}{c}\left(\frac{dv}{dt}\right)^2 + \frac{f}{2dc}v^2\frac{dv}{dt} = 0 \qquad (2\text{-}16)$$

with initial conditions:

$$t = 0 \qquad v = 0$$

$$t = 0 \qquad \frac{dv}{dt} = \frac{P_o A_p}{M_p} = \frac{P_o}{\rho_o \ell_p}$$

If we write $q = dv/dt$, Eq. (2-16) now becomes

$$\frac{dq}{dv} + \frac{1}{c} q + \frac{f}{2dc} = 0 \tag{2-17}$$

The integration of Eq. (2-17) by assuming an average f with boundary condition

$$v = 0, \; q = \frac{dv}{dt} = \frac{P_o}{\rho_p A_p}$$

is

$$\frac{dv}{dt} = q = \frac{P_o}{C_p \ell_p} \exp\left(-\frac{v}{c}\right) - \frac{f}{2dc}\left[v^2 c - 2vc^2 + 2c^3 + 2c^3 \exp\left(-\frac{v}{c}\right)\right] \tag{2-18}$$

Written in dimensionless form, it is (for the isothermal case)

$$\frac{d\bar{v}}{d\bar{t}} = \frac{2}{K+1}\bar{p} = F(\bar{v},K,f') = \frac{2}{(K+1)} \exp(-K^{1/2}\,\bar{v})$$

$$- \frac{1}{(K+1)} f' \left\{K\bar{v}^2 - 2K^{1/2}\bar{v} + 2\left[1-\exp(-K^{1/2}\bar{v})\right]\right\} \tag{2-19}$$

For $f' = 0$ it is reduced to $\bar{p} = \exp(-K^{1/2}\bar{v})$ as expected for the ideal isothermal case.

Here

$$\bar{v} = \frac{v}{c_K} \qquad c_K = (K\ R\ T_o)^{1/2} \qquad K = \frac{c_p}{c_v} = 1.4 \quad (\text{for } H_2)$$

$$\bar{p} = \frac{p}{p_o}, \quad f' = f \cdot \frac{\rho_p \ell_p}{\rho_o d}$$

$$\bar{t} = \frac{t}{\tau} \qquad \tau = \frac{(K+1)}{2} \frac{p_o}{\rho_p \ell_p}$$

The results of Eq. (2-19) are expressed in Figures 4 and 5 to show the relationship between pressure and pellet velocity, and the maximum possible pellet velocity (where $p/p_o = 0$) with different friction coefficients. Further detailed information could be obtained by numerical integration of Eq. (2-19) in the following form

$$\bar{t} = \int_o^{\bar{t}} d\bar{t} = \int_o^{\bar{v}} \frac{d\bar{v}}{F(\bar{v},K,f')} \tag{2-20}$$

The frictional coefficient of a fully developed viscous flow through a round tube is a function of the Reynolds number ($= \rho v d/\mu$) and roughness of the tube[4]. A typical value is about 0.01-0.03 for turbulent flow happening herein. Here we assume an average frictional coefficient of fully developed flow instead of one for unsteady non-fully developed flow. The latter usually gives a somewhat underestimated value of the frictional loss, though the deviation is believed to be small.

4. HEAT TRANSFER BETWEEN PROPELLENT GAS AND BARREL

From the ideal theory of single-stage or constant base pressure acceleration pellet injection[1], it was clear that raising the initial gas temperature gives favoured results in both cases for adiabatic flow. Here a simplified model is proposed to re-examine the heat transfer rate between the gas and wall to show the seriousness of the deviation from the adiabatic case; some conclusions emerging may change the original expectations.

In spite of the complex formulation of the heat transfer between propellent gas and barrel in a real pneumatic pellet injector owing to unsteady and variable flow properties along the barrel, a model is proposed to give a rough estimate of this problem in an indirect way. Consider a high-temperature (T_g) gas passing through a round tube with the wall at an initial temperature, $T_w = T_{wo}$ ($T_g > T_{wo}$. The transit time τ is of order of one millisecond. (For modern D_2-pellet injectors,

$$\tau \sim \frac{\text{barrel length } L}{\text{average gas velocity } v_{av}} \sim \frac{1 \text{ m}}{10^3 \text{ m/s}} \sim 10^{-3} \text{ sec})$$

When $t > 0$, the calculated wall temperature starts to rise as it would in an unsteady heat conduction problem with forced convection in the gas side. Since time is short, the energy deposition layer in the wall is rather thin ($\delta \sim (K_w/\rho_w C_w\, t)^{1/2} \sim 1 \times 10^{-2}$ cm for stainless steel barrel) as compared to the wall thickness as well as the wall average diameter. The use of a semi-infinite planar model instead of a finite thickness cylindrical model provides sufficient accuracy.

It is further assumed that the gas temperature remains constant along the barrel. It means energy should be supplied to the gas to compensate for its loss into the barrel. When actually no heat addition is provided in a real case, the model simply could be interpreted in certain cases as follows: when the "energy input" to the wall is much greater than the gas enthalpy itself as estimated here, the gas enthalpy will be dropped to nearly that corresponding to the wall temperature. For the opposite case, if "energy input" is much less than the gas enthalpy, this means that the condition is close to adiabatic. This is an indirect proof of the problem in order to simplify the calculations.

The convective heat transfer from gas to wall can be expressed as

$$q = \alpha(T_g - T_{w_{x=0}}) \qquad (3\text{-}1)$$

Here the convective heat transfer coefficient is determined by[6]

$$Nu = 0.023\ (Re)^{0.8}\ (Pr)^{0.3} \qquad (3\text{-}2)$$

With

$$Nu = \frac{\alpha d}{K_g}, \quad Re = \frac{\rho_g v_g d}{\mu_g}, \quad Pr = \frac{\mu_g c_{p,g}}{K_g}$$

The temperature distribution in the wall can be determined by

$$\frac{\partial T_w}{\partial t} = \frac{K_w}{\rho_w c_{h,w}} \frac{\partial^2 T_w}{\partial x^2} \qquad (3\text{-}3)$$

with initial and boundary conditions

$$t = 0,\ x > 0 \qquad T_w = T_{wo}$$

$$t > 0 \quad x = 0 \qquad -K_w \left.\frac{\partial T}{\partial x}\right|_{x=0} = \alpha\ T_g - (T_w)_{x=0}$$

the relation is[7]

$$Nu = 0.023\ (Re)^{0.8}\ (Pr)^{0.3}$$

$$= 724$$

$$\alpha = Nu\ \frac{K_g}{d} = 724 \times \frac{0.63 \times 10^{-2}}{0.4} = 11.40\ w/cm^2\ ^oK$$

$$h = \frac{\alpha}{K_w} = \frac{11.4}{0.2} = 57\ cm^{-1}$$

$$Z = h\sqrt{D_w t}$$

$$= 57 \times \sqrt{0.0414} \times 10^{-3} = 0.366$$

$$\frac{Q_w}{Q_g} = \frac{8 \times 7.7 \times 0.627}{0.8 \times 10^{-3} \times 16.15}\ \frac{\sqrt{0.0414} \times 10^{-3}}{0.4} \times 0.18$$

$$= 4$$

Since $Q_w/Q_g >> 1$, as mentioned before, it means that heat transfer between the gas and the wall is so serious that if a metal wall such as stainless steel were used, the gas enthalpy would be nearly completely lost into the wall within a fraction of the barrel length and then the gas temperature will remain at near the wall at the initial value in the remaining portion of the barrel. The effective heat conductivity is here

$$K_{eff} = \frac{\alpha d}{2} = 11.40 \times 0.2 = 2.28\ w/cm\text{-}^oK$$

and is much higher than the molecular heat conductivity K_g by at least two orders of magnitude, apparently due to a turbulent effect. It is also higher than the heat conductivity of the metal wall by one order of magnitude. The energy deposited thickness of the wall can be estimated from unsteady heat conduction equation as

$$\delta_w = (D_w t)^{1/2} = (0.0414 \times 10^{-3})^{1/2}$$

$$= 0.64 \times 10^{-2} \text{ cm}$$

The effective energy conductive thickness of the gas is

$$\delta_g = \left(\frac{K_{g,eff}}{\rho_g c_{h,g}} t\right)^{1/2} = \left(\frac{2.28}{1.6 \times 10^{-3} \times 16.15} \times 10^{-3}\right)^{1/2}$$
$$= 0.3 \text{ cm}$$

still larger than the barrel radius. It indicates that we may consider quasi-steady heat convection in gas side to be relatively reasonable.

A rougher estimate may more clearly show how important is the value of heat transfer. From the unsteady heat conduction equation in the wall

$$\frac{\partial T}{\partial t} = \frac{K_{h,w}}{\rho_w c_w} \frac{\partial^2 T}{\partial x^2}$$

The energy-deposited thickness is roughly

$$\delta = (D_w t)^{1/2}$$

Assuming a linear distribution of temperature in the wall this shows

$$\frac{Q_w}{Q_g} = \frac{\text{energy deposited into the wall}}{\text{energy content of the gas}}$$

$$= \frac{\rho_w c_w \pi d L \delta \frac{\Delta T}{2}}{\rho_g c_g \frac{\pi}{4} d^2 L \Delta T} = \frac{\rho_w c_w}{\rho_g c_g} \cdot \frac{2\delta}{d}$$

or

$$\frac{Q_w}{Q_g} = \frac{\rho_w c_w}{\rho_g c_g} \frac{2\delta}{d} T ,$$

if initial pressure remains constant. The higher the initial temperature, the greater is Q_w/Q_g is. If $T_0 = 3000^{\circ}K$, it will have Q_w/Q_g nearly ten times as compared to that at room temperature.

With the parameters given in the example above, it is shown that

$$\frac{Q_w}{Q_g} = \frac{7.7 \times 0.627 \times 2 \times (0.0414 \times 10^{-3})^{1/2}}{1.6 \times 10^{-3} \times 16.15 \times 0.4}$$

$$= 6$$

5. DISCUSSIONS AND SUGGESTIONS

1. Convective heat transfer is so strong between the high-temperature propellent gas (H_2) and the metal wall that most of the gas enthalpy seems to be lost within a fraction of the barrel length. Therefore, in these cases the gas to be heated in the chamber before the barrel has a very limited effect on the performance of the injector at least from the point of view of the rising gas temperature. (The pressure increase in the chamber caused by the heating of the gas has its own effect on the injector).

2. From the above discussion, the following is suggested to improve the performance of next generation high-speed pneumatic injectors: (a) either the use of a heat-insulated barrel (or with a thin heat-insulated film deposited on the inner wall of metal barrel) with heated gas at the entrance (near adiabatic case), (b) use of heated metal barrel (composed of for example, Mo or Ta, to be heated to ~ 2000°K) with non-heated gas at the entrance (near isothermal case corresponding to the wall temperature).

3. Though the frictional loss between the pellet and wall seems as if it could be neglected, that between the propellent and the wall could have a significant effect on high-speed pellet injector performance. Simplified calculational procedures are suggested to take account of this effect in certain cases for both single-stage and constant base pressure acceleration.

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NOMENCLATURE

a	acceleration
A	area
C	sound velocity
c_h	specific heat
d	diameter
D	heat diffusity
f	friction coefficient
G	gas flow rate
K	heat conductivity; specific heat ratio
ℓ	length
L	length
M	mass
n	polytropic factor
Nu	Nusselt number
p	pressure
Pr	Prandtl number
q	heat transfer (unit area)
Q	total heat transfer rate
R	gas constant
Re	Reynolds number
t	time
v	velocity
ρ	density
μ	viscosity
α	convective heat transfer coefficient
δ	thickness

Subscript:

b	base of pellet; barrel
f	fluid
g	gas
p	pellet
q	dimensionless form of q
w	wall

- initial drive gas state for single-stage driven case or state of gas at entrance of barrel for constant base pressure acceleration

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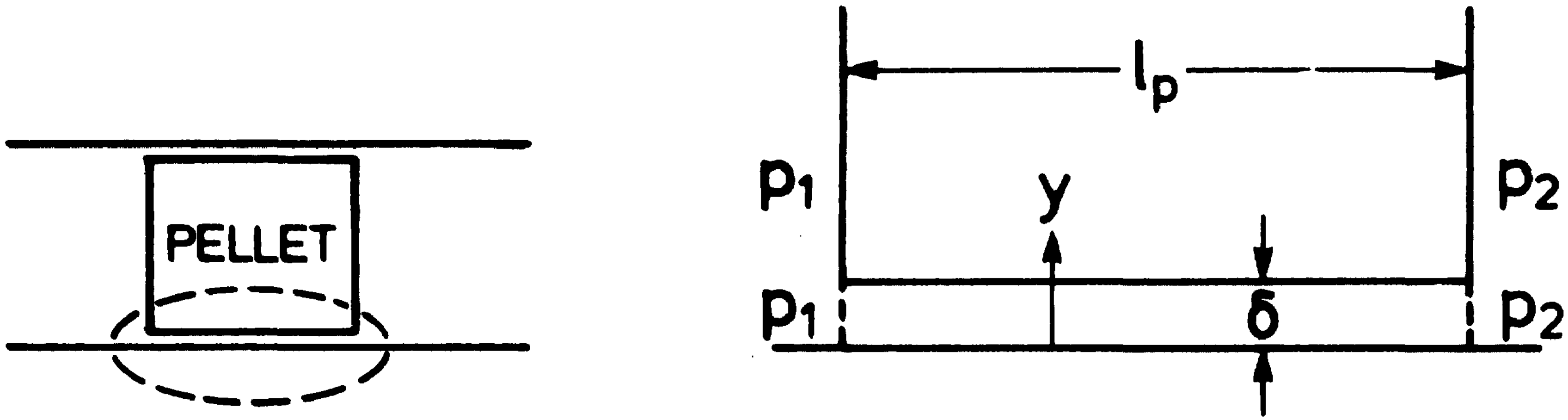

Fig. 1. Schematic diagram for pellet-wall interaction consideration.

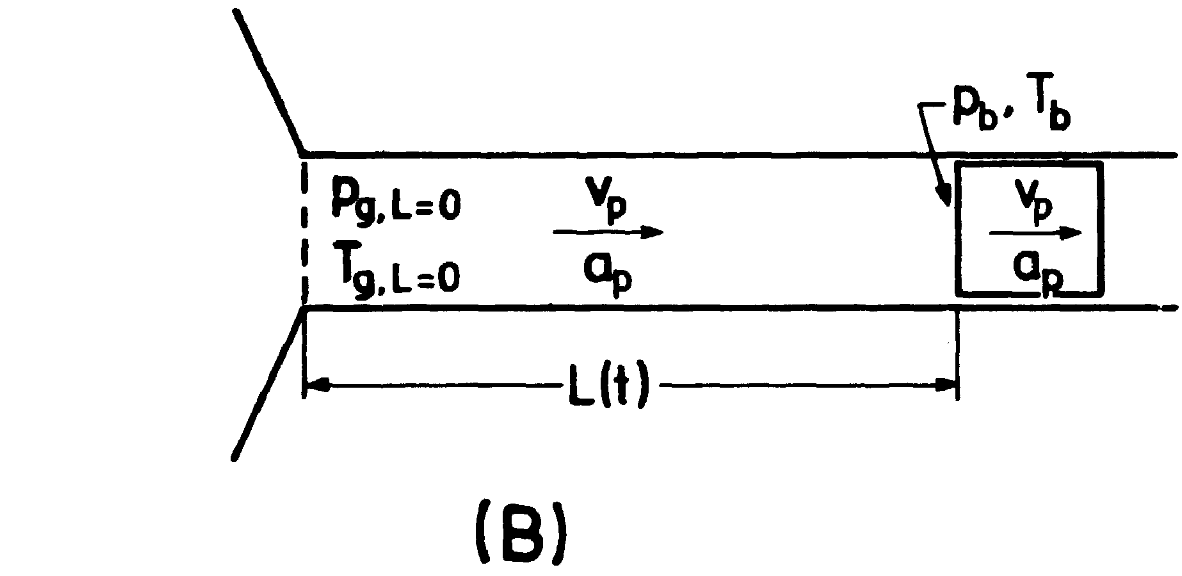

Fig. 2 A. Simple single-stage pellet injector.

B. Constant base pressure acceleration pellet injector.

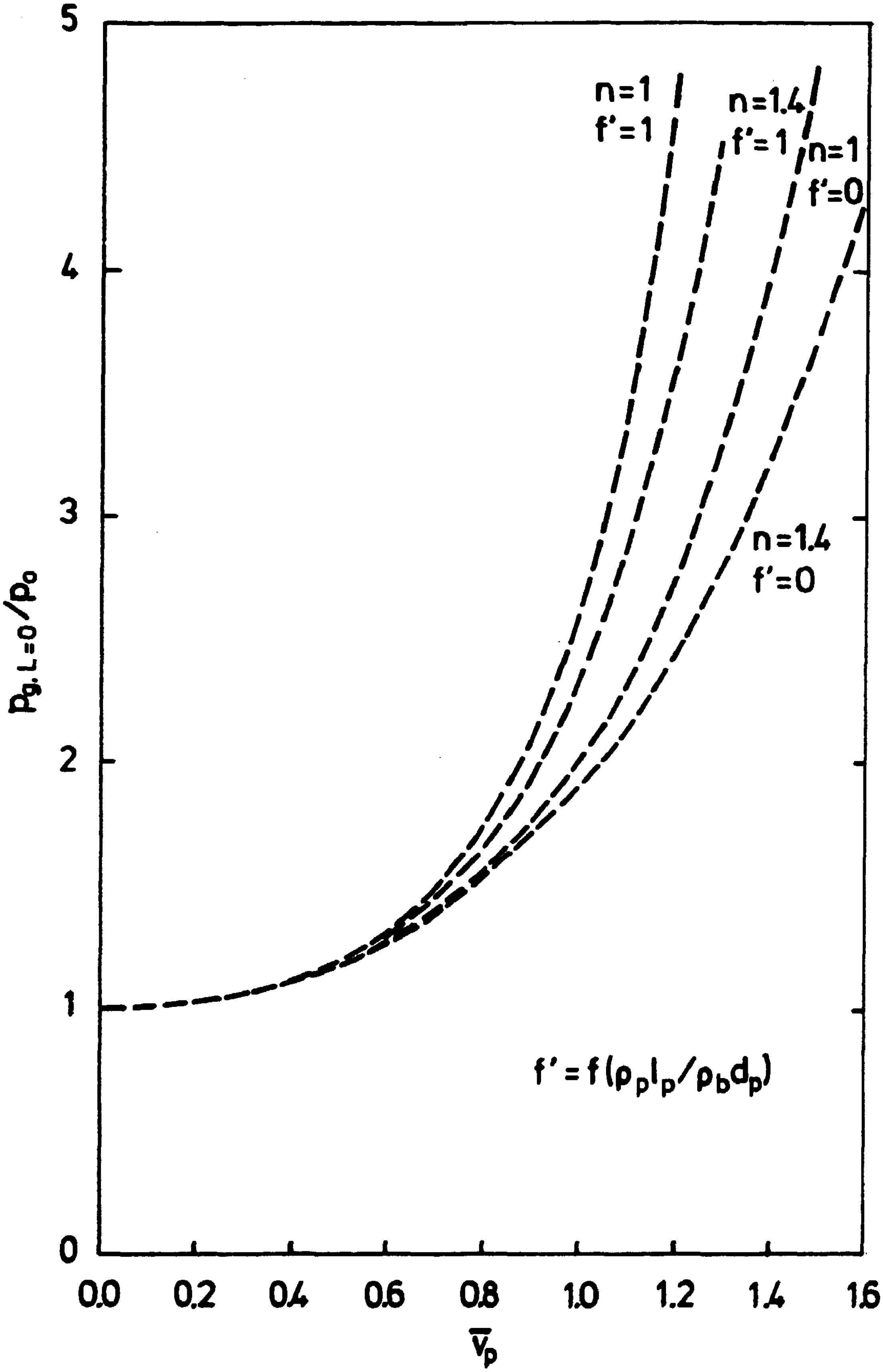

Fig. 3. Static pressure programming for constant base pressure acceleration.

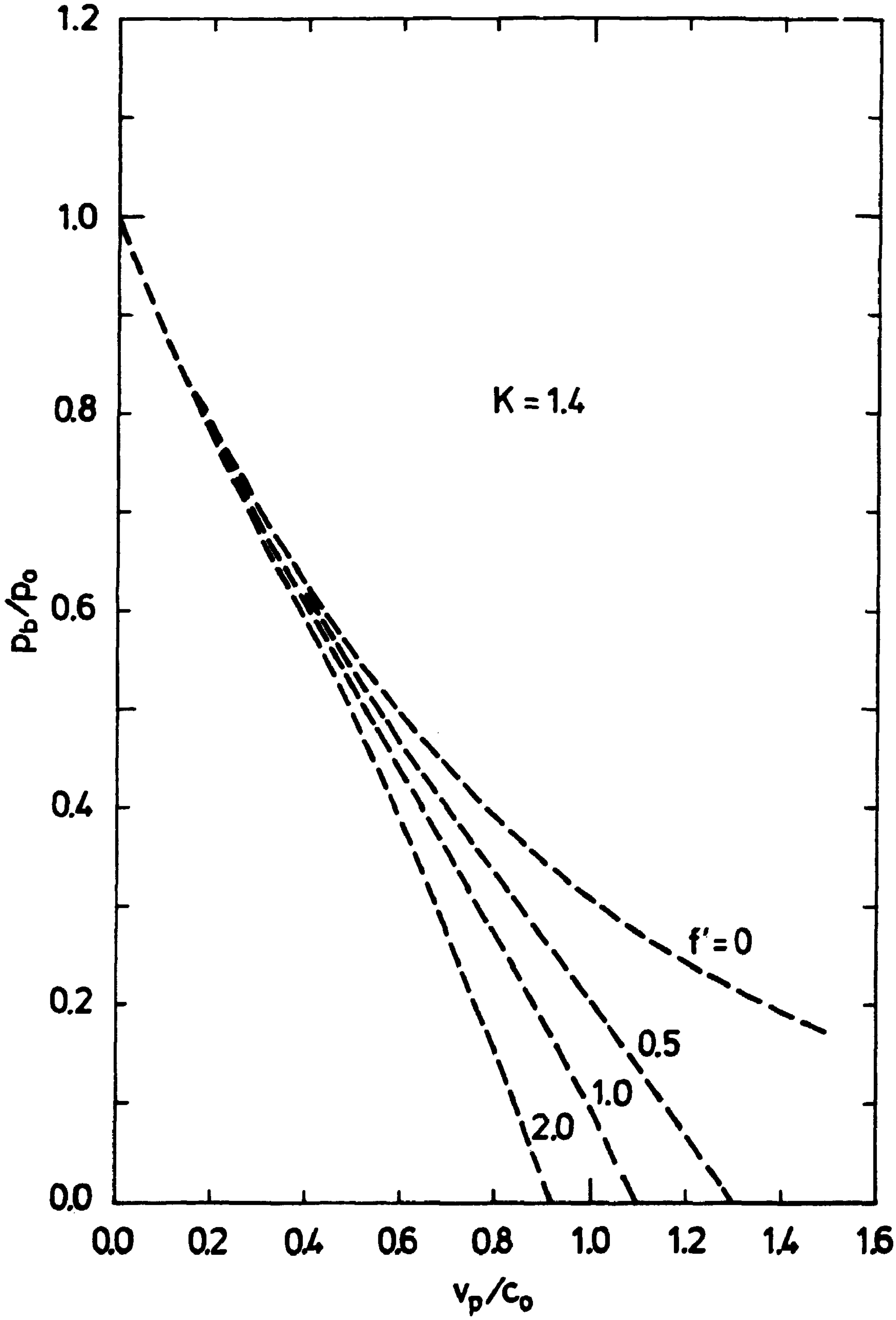

Fig. 4. Base pressure - pellet velocity relationship for simple single stage injectors (isothermal case with friction).

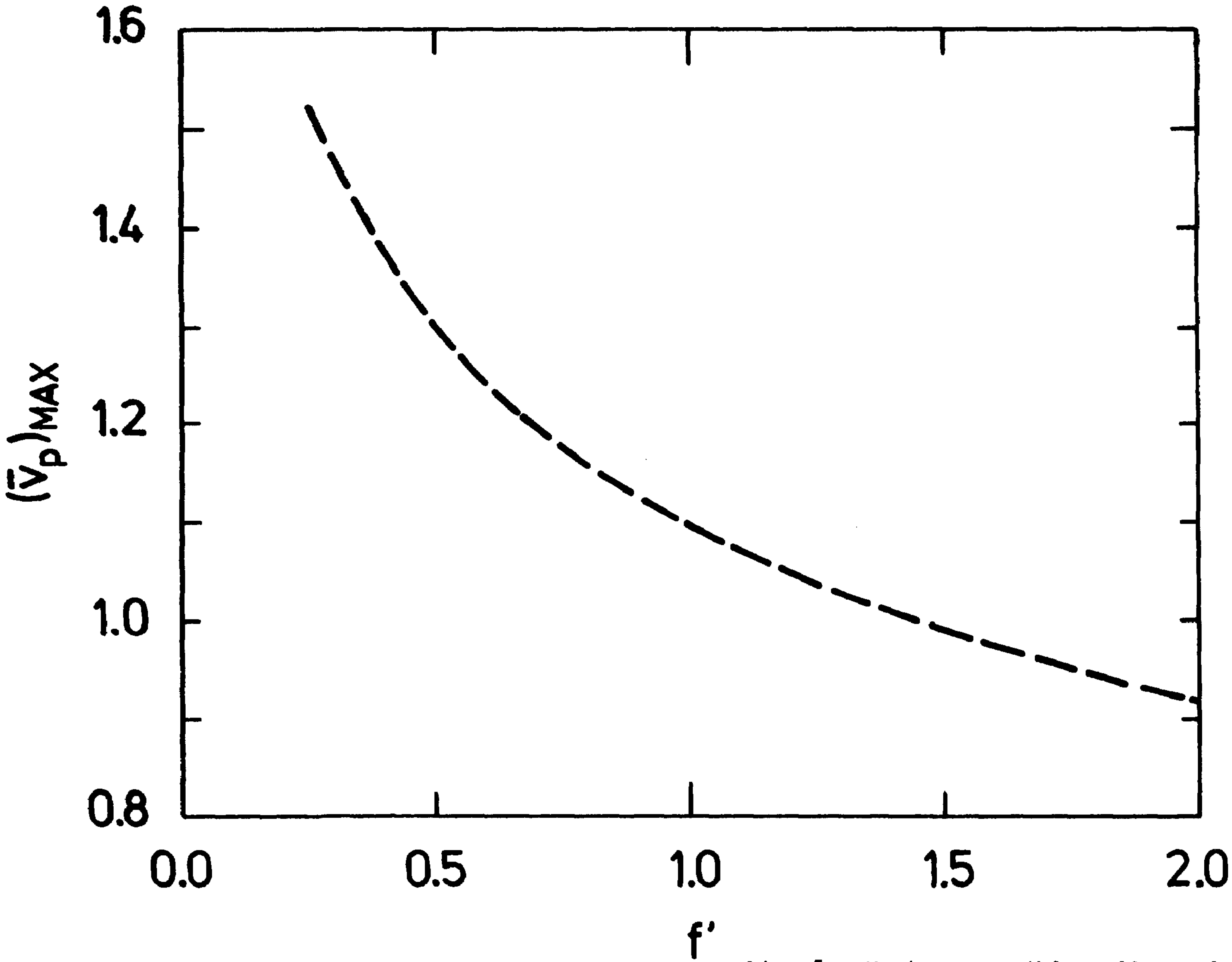

Fig. 5. Maximum possible pellet velocity vs. friction coefficient for simple single-stage injectors (isothermal case).

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Several real effects that take place in a pneumatic pellet injector are examined. These are the heat transfer between a high-temperature propellent gas and the metal wall of the injector, and the frictional loss between the propellent and wall.

Descriptors - INIS

FRICTION; HEAT TRANSFER; PELLET INJECTION; PNEUMATIC TRANSPORT; TOKAMAK DEVICES

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